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Energy and exergy analyses of thermal power plants: A review

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ABSTRACT

The energy supply to demand narrowing down day by day around the world, the growing demand of power has made the power plants of scientific interest, but most of the power plants are designed by the energetic performance criteria based on first law of thermodynamics only. The real useful energy loss cannot be justified by the fist law of thermodynamics, because it does not differentiate between the quality and quantity of energy. The present study deals with the comparison of energy and exergy analyses of thermal power plants stimulated by coal and gas. This article provides a detailed review of different studies on thermal power plants over the years. This review would also throw light on the scope for further research and recommendations for improvement in the existing thermal power plants.

Contents

1. Introduction

Energy consumption is one of the most important indicator showing the development stages of countries and living standards of communities. Population increment, urbanization, industrializing, and technologic development result directly in increasing energy consumption. This rapid growing trend brings about the crucial environmental problems

such as contamination and greenhouse effect. Currently, 80% of electricity in the world is approximately produced from fossil fuels (coal, petroleum, fuel-oil, natural gas) fired thermal power plants, whereas 20% of the electricity is compensated from different sources such as hydraulic, nuclear, wind, solar, geothermal and biogas [1]. Generally, the performance of thermal power plants is evaluated through energetic performance criteria based on first law of thermodynamics, including electrical power and thermal efficiency. In recent decades, the exergetic performance based on the second law of thermodynamics has found as useful method in the design, evaluation, optimization and improvement of thermal power plants. The exergetic performance analysis can not only determine magnitudes, location and causes of irreversibilities in the plants, but also provides more meaningful assessment of plant individual components efficiency. These points of the exergetic performance analysis are the basic differences from energetic

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Nomenclature

$\dot{m}$	mass flow rate kg/s
T	temperature (°C)
$\dot{W}$	work done (kJ/s)
$\dot{Q}$	heat transfer (kJ/s)
E	flow energy (kJ/s)
Ex^{Q}	exergetic heat (kJ/s)
Ψ	flow exergy (kJ/s)
S	specific entropy (kJ/kg K)
h	Specific entropy (kJ/kg)
$\dot{I}_{\mathrm{destroyed}}$	irreversibility (kJ/s)

Subscripts

a	air
f	fuel
g	gas
o	dead state
s	steam
w	water
I	inlet
O	outlet

performance analysis. Therefore, it can be said that performing exergetic and energetic analyses together can give a complete depiction of system characteristics. Such a comprehensive analysis will be a more convenient approach for the performance evaluation and determination of the steps towards improvement [4–6]. In the literature, there exist a number of papers concerning energetic and exergetic performances of coal-fired thermal power plants. For instance, Hasan et al. [1] presented thermodynamic inefficiencies as well as reasonable comparison of each plant to others are identified and discussed for the coal-fired thermal power plants in Turkey.

Aljundi [2] determined the performance of the plant was estimated by a component wise modeling and a detailed break-up of energy and exergy losses for the considered steam power plant in Jordan. Datta et al. [3] presented exergy analysis of a coal-based thermal power plant by splitting up the entire plant cycle into three zones for the analysis. Naterer et al. [4] analyzed the coal-fired thermal power plant with measured boiler and turbine losses. Rosen [5] presented energy and exergy-based comparison of coal-fired and nuclear steam power plants. Ganapathy et al. [6] determined the energy losses and the exergy losses of the individual components of the lignite fired thermal power plant. Zubair and Habib [7] performed second law based thermodynamic analysis of the regenerative-reheat Rankine cycle power plants. Reddy and Butcher [8] analyzed waste heat recovery based power generation system based on second law of thermodynamics. Suresh et al. [9] determined the exergetic performance of the coal-based thermal power plants using subcritical, supercritical, and ultra-supercritical steam conditions. Oktay [10] presented exergy loss and proposed improving methods for a fluidized bed power plant in Turkey.

Reddy and Mohamed [11] analyzed a natural gas fired combined cycle power generation unit to investigate the effect of gas turbine inlet temperature and pressure ratio on the exergetic efficiency. Srinivas et al. [12] analyzed the combined cycle power plant using methane as a fuel using the first and second law of thermodynamics. Can and Smith [13] described a simple but accurate method to estimate the Rankine bottoming cycle power output directly from the exergy of the gas turbine exhaust, utilizing the second law of thermodynamics. Datta et al. [14] presented energy and exergy analyses of an externally fired gas turbine cycle integrated with biomass gasifier for distributed power generation. Sue and Chuang [15] performed the engineering design and theoretical exergetic analysis of the combustion gas turbine based power generation systems. Bilgen [16] presented the exergetic and engineering analyses as well as a simulation of gas turbine-based cogeneration plants consisting of a gas turbine, heat recovery steam generator and steam turbine. Khaliq and Kaushik [17] presented the second-law approach for the thermodynamic analysis of the reheat combined Brayton/Rankine power cycle. Woudstra et al. [18] determined the cogeneration process, levels of steam generation to reduce the heat transfer losses in the heat recovery steam generator (HRSG) and the exergy loss due to the exhaust of flue gas to the stack.

Keeping in view the facts stated above, it can be expected that performing an analysis based on the same definition of performance criteria will be meaningful for performance comparisons, assessments and improvement for thermal power plants. Additionally, considering both the energetic and exergetic performance criteria together can guide the ways of efficient and effective usage of fuel resources by taking into account the quality and quantity of the energy used in the generation of electric power in thermal power plants. The purpose of this study presented here is to carry out energetic and exergetic performance analyses, at the design conditions, for the existing coal and gas-fired thermal power plants in order to identify the needed improvement. For performing this aim, we summarized thermodynamic models for the considered power plants on the basis of mass, energy and exergy balance equations. The thermodynamic model simulation results are compared. In the direction of the comprehensive analysis results, the requirements for performance improvement are evaluated.

2. Energy and exergy analyses of coal fired power plants

Coal based thermal power plant generally operates on Rankine cycle. In ideal vapor power cycle many, such as Carnot cycle has impracticalities associated with it can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser [19].

2.1. Description of coal fired power plant

We are considering the analysis of a cumulative coal fired thermal power plant with all methods of the efficiency increasing technics like lowering the condenser pressure, superheating the steam to high temperatures, increasing the boiler pressure, reheat and regenerative Rankine cycle, as shown in Fig. 1. A continuous mass flow diagram for one unit of the power plant modeled in this study includes the main components such as high, intermediate and low pressure turbines (HPT, IPT and LPT), a boiler (B), number of pumps (P), a dearetor (D), a generator (G), a condenser (C), low and high pressure feed water heaters (LPH and HPH). The thermodynamic models are based on fundamental mass and energy balances. Using the energy and mass balance equations for each component in the power plant model, it is possible to compute energy and exergy contents in terms of turbine power outputs, pump power consumptions, boiler heat requirements, energy and exergy flows at each node of the plants, component first and second efficiencies, component irreversibilities in the plants, and so on.

2.2. Energy analysis

In an open flow system there are three types of energy transfer across the control surface namely working transfer, heat transfer, and energy associated with mass transfer and/or flow. The first law of thermodynamics or energy balance for the steady flow process

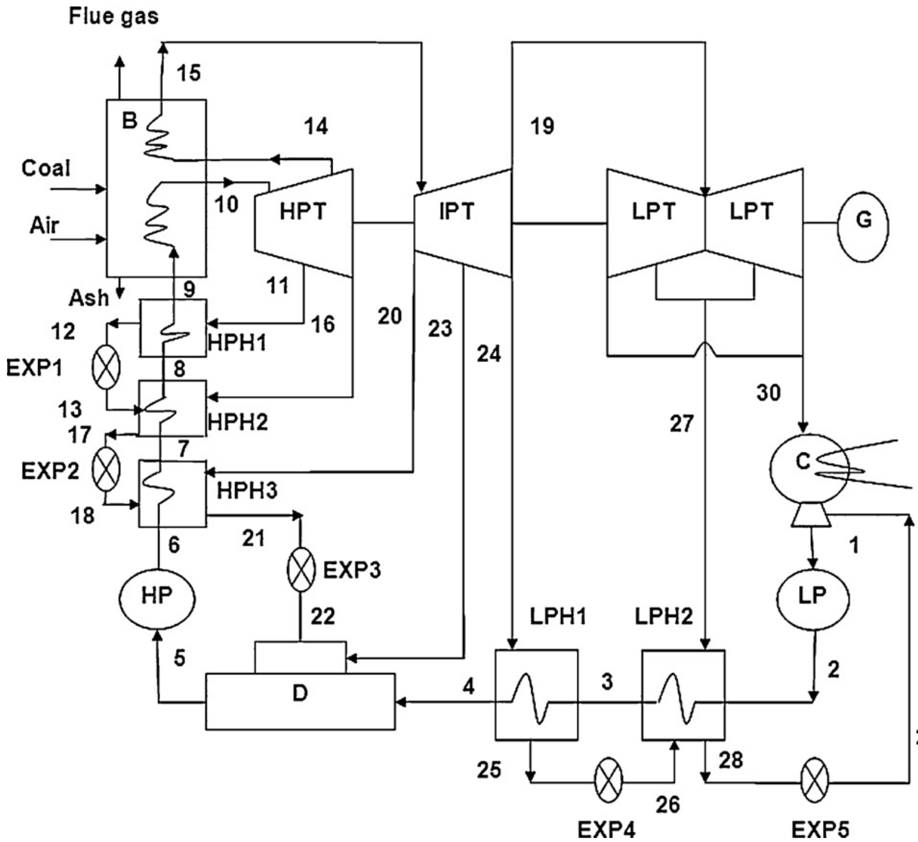

Fig. 1. Coal-fired thermal power plant.

of an open system is given by:

$$\sum \dot{Q}_k + \dot{m}\left(h_i + \frac{C_i^2}{2} + gZ_i\right) = \dot{m}\left(h_o + \frac{C_o^2}{2} + gZ_o\right) + \dot{W}$$

where Q_k heat transfer to system from source at temperature T_k, and W is the net work developed by the system. The other notations C is the bulk velocity of the working fluid, Z, is the altitude of the stream above the sea level, g is the specific gravitational force.

The energy or first law efficiency η_{I} of a system and/or system component is defined as the ratio of energy output to the energy input to system/component i.e.

$$\eta_{\mathrm{I}} = \frac{\text{Desired output energy}}{\text{Input energy supplied}}$$

To analyze the possible realistic performance, a detailed energy analysis of the coal fired thermal power plant system has been carried out by ignoring the kinetic and potential energy change

(a) The energy balance for boiler:

The energy balance for the combustion/boiler is give by:

$$0 = Q_k - \dot{m}_w(h_{10} - h_9) - \dot{m}_s(h_{15} - h_{14})] - \text{Energy loss}$$

where m_w is the mass flow rate of water, m_s is the mass flow rate of steam combustion which gives:

$$\text{Energy loss} = Q_k - \dot{m}_w(h_{10} - h_9) - \dot{m}_s(h_{15} - h_{14})$$

The first law efficiency is defined as

$$\eta_{\mathrm{I,Boiler}} = \frac{\text{Energy output}}{\text{Energy input}} = 1 - \frac{\text{Energy loss}}{\text{Energy input}} = \frac{\dot{m}_w(h_{10} - h_9) + \dot{m}_s(h_{15} - h_{14})}{\dot{Q}_k}$$

(b) The energy balance for the high pressure turbine is give by:

$$W_{\mathrm{HPT}} = \dot{m}_{10}(h_{10} - h_{11}) + (\dot{m}_{10} - \dot{m}_{11})(h_{11} - h_{14}) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_{10}(h_{10} - h_{11}) + (\dot{m}_{10} - \dot{m}_{11})(h_{11} - h_{14}) - W_{\mathrm{HPT}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,HPT}} = 1 - \frac{\text{Energy loss}}{\dot{m}_{10}(h_{10} - h_{11}) + (\dot{m}_{10} - \dot{m}_{11})(h_{11} - h_{14})} = \frac{W_{\mathrm{HPT}}}{\dot{m}_{10}(h_{10} - h_{11}) + (\dot{m}_{10} - \dot{m}_{11})(h_{11} - h_{14})}$$

(c) The energy balance for the intermediate pressure turbine is give by:

$$W_{\mathrm{IPT}} = \dot{m}_{15}(h_{15} - h_{20}) + (\dot{m}_{15} - \dot{m}_{20})(h_{20} - h_{23}) + (\dot{m}_{15} - \dot{m}_{20} - \dot{m}_{23})(h_{23} - h_{19}) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_{15}(h_{15} - h_{20}) + (\dot{m}_{15} - \dot{m}_{20})(h_{20} - h_{23}) + (\dot{m}_{15} - \dot{m}_{20} - \dot{m}_{23})(h_{23} - h_{19}) - W_{\mathrm{IPT}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,IPT}} = 1 - \frac{\text{Energy loss}}{\dot{m}_{15}(h_{15}-h_{20}) + (\dot{m}_{15}-\dot{m}_{20})(h_{20}-h_{23}) + (\dot{m}_{15}-\dot{m}_{20}-\dot{m}_{23})(h_{23}-h_{19})}$$
$$= \frac{W_{\mathrm{IPT}}}{\dot{m}_{15}(h_{15}-h_{20}) + (\dot{m}_{15}-\dot{m}_{20})(h_{20}-h_{23}) + (\dot{m}_{15}-\dot{m}_{20}-\dot{m}_{23})(h_{23}-h_{19})}$$

(d) The energy balance for the low pressure turbine is give by:

$$W_{\mathrm{LPT}} = \dot{m}_{19}(h_{19}-h_{27}) + (\dot{m}_{19}-\dot{m}_{27})(h_{27}-h_{30}) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_{19}(h_{19}-h_{27}) + (m_{19}-m_{27})(h_{27}-h_{30}) - W_{\mathrm{LPT}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,LPT}} = \frac{\text{Energy loss}}{\dot{m}_{19}(h_{19}-h_{27}) + (\dot{m}_{19}-\dot{m}_{27})(h_{27}-h_{30})} = \frac{W_{\mathrm{LPT}}}{\dot{m}_{19}(h_{19}-h_{27}) + (\dot{m}_{19}-\dot{m}_{27})(h_{27}-h_{30})}$$

Condenser sub system

(e) The energy balance for the condenser is give by:

$$0 = \dot{m}_{30}(h_{30}-h_1) - Q_k - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_{30}(h_{30}-h_1) - Q_k$$

The first law efficiency is:

$$\eta_{\mathrm{I,Cond}} = 1 - \frac{\text{Energy loss}}{\dot{m}_{30}(h_{30}-h_1)}$$

Pump sub system

(f) The energy balance for the low pressure pump is give by:

$$-W_{\mathrm{LPP}} = \dot{m}_1(h_1-h_2) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_1(h_1-h_2) + W_{\mathrm{LPP}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,LPP}} = 1 - \frac{\text{Energy loss}}{W_{\mathrm{LPP}}} = \frac{m_1(h_2-h_1)}{W_{\mathrm{LPP}}}$$

(g) The energy balance for the high pressure pump is give by:

$$-W_{\mathrm{HPP}} = \dot{m}_5(h_5-h_6) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_5(h_5-h_6) + W_{\mathrm{HPP}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,HPP}} = 1 - \frac{\text{Energy loss}}{W_{\mathrm{HPP}}} = \frac{\dot{m}_1(h_6-h_5)}{W_{\mathrm{HPP}}}$$

Feed water heater sub system

(h) The energy flow equation for the high presure feed water heater (HPH1) system becomes:

$$0 = \dot{m}_{11}(h_{11}-h_{12}) - \dot{m}_8(h_9-h_8) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = [\dot{m}_{11}(h_{11}-h_{12}) - \dot{m}_8(h_9-h_8)]$$

The first law efficiency is:

$$\eta_{\mathrm{I,HPH1}} = 1 - \frac{\text{Energy loss}}{\dot{m}_{11}(h_{11}-h_{12})} = \frac{\dot{m}_8(h_9-h_8)}{\dot{m}_{11}(h_{11}-h_{12})}$$

(i) The energy flow equation for the high presure feed water heater (HPH2) system becomes:

$$0 = \dot{m}_{16}(h_{16}-h_{17}) - \dot{m}_7(h_8-h_7) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = [\dot{m}_{16}(h_{16}-h_{17}) - \dot{m}_7(h_8-h_7)]$$

The first law efficiency is:

$$\eta_{\mathrm{I,HPH2}} = 1 - \frac{\text{Energy loss}}{\dot{m}_{16}(h_{16}-h_{17})} = \frac{\dot{m}_7(h_8-h_7)}{\dot{m}_{16}(h_{16}-h_{17})}$$

(j) The energy flow equation for the high presure feed water heater (HPH3) system becomes:

$$0 = \dot{m}_{20}(h_{20}-h_{21}) - \dot{m}_6(h_7-h_6) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = [\dot{m}_{20}(h_{20}-h_{21}) - \dot{m}_6(h_7-h_6)]$$

The first law efficiency is:

$$\eta_{\mathrm{I,HPH3}} = 1 - \frac{\text{Energy loss}}{\dot{m}_{20}(h_{20}-h_{21})} = \frac{\dot{m}_6(h_7-h_6)}{\dot{m}_{20}(h_{20}-h_{21})}$$

(k) The energy flow equation for the low presure feed water heater (LPH1) system becomes:

$$0 = \dot{m}_{24}(h_{24}-h_{25}) - \dot{m}_3(h_4-h_3) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = [\dot{m}_{24}(h_{24}-h_{25}) - \dot{m}_6(h_4-h_3)]$$

The first law efficiency is:

$$\eta_{I,LPH1} = 1 - \frac{\text{Energy loss}}{\dot{m}_{24}(h_{24}-h_{25})} = \frac{\dot{m}_3(h_4-h_3)}{\dot{m}_{24}(h_{24}-h_{25})}$$

(l) The energy flow equation for the low presure feed water heater (LPH2) system becomes:

$$0 = \dot{m}_{27}(h_{27}-h_{28}) - \dot{m}_2(h_3-h_2) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = [\dot{m}_{27}(h_{27}-h_{28}) - \dot{m}_3(h_4-h_3)]$$

The first law efficiency is:

$$\eta_{II,LPH2} = 1 - \frac{\text{Energy loss}}{\dot{m}_{27}(h_{27}-h_{28})} = \frac{\dot{m}_2(h_3-h_2)}{\dot{m}_{27}(h_{27}-h_{28})}$$

(m) Dearetor sub system

It is an adiabatic mixing chamber where a hot streams 23, 22 are mixed with a cold stream 4, forming a mixture 5, the energy supplied is the sum of the energies of the hot and cold streams, and the energy recovered is the energy of the mixture. The energy flow equation for the dearetor system becomes:

$$0 = \dot{m}_{22}h_{22} + \dot{m}_{23}h_{23} + \dot{m}_4h_4 - \dot{m}_5h_5 - \text{Energy loss}$$

where $\dot{m}_5 = \dot{m}_{22} + \dot{m}_{23} + \dot{m}_{24}$. This gives:

$$\text{Energy loss} = \dot{m}_{22}h_{22} + \dot{m}_{23}h_{23} + \dot{m}_4h_4 - \dot{m}_5h_5$$

The first law efficiency is:

$$\eta_{II,Der} = 1 - \frac{\text{Energy loss}}{\dot{m}_{22}h_{22} + \dot{m}_{23}h_{23} + \dot{m}_4h_4} = \frac{\dot{m}_5h_5}{\dot{m}_{22}h_{22} + \dot{m}_{23}h_{23} + \dot{m}_4h_4}$$

2.3. Exergy analysis

Exergy is a generic term for a group of concepts that define the maximum possible work potential of a system, a stream of matter and/or heat interaction; the state of the (conceptual) environment being used as the datum state. In an open flow system there are three types of energy transfer across the control surface namely working transfer, heat transfer, and energy associated with mass transfer and/or flow. The work transfer is equivalent to the maximum work, which can be obtained from that form of energy. The exergy (Ψ_Q) of heat transfer Q from the control surface at temperature T is determined from maximum rate of conversion of thermal energy to work W_{max}. is given by:

$$W_{max} = \Psi_Q = Q\left(1 - \frac{T_0}{T}\right)$$

Exergy of steady flow stream of matter is the sum of kinetic, potential and physical exergy. The kinetic and potential energy are almost equivalent to exergy. The physical specific exergy Ψ_i and Ψ_o depends on initial state of matter and environmental state. Energy analysis is based on the first law of thermodynamics, which is related to the conservation of energy. Second law analysis is a method that uses the conservation of mass and degradation of the quality of energy along with the entropy generation in the analysis design and improvement of energy systems. Exergy analysis is a useful method; to complement but not to replace energy analysis.

The exergy flow for steady flow process of an open system is given by

$$\sum\left(1-\frac{T_0}{T_k}\right)Q_k + \sum_{in}\dot{m}\Psi_i = \Psi_W + \sum_{out}\dot{m}\Psi_o + \dot{I}_{destroyed};$$

$$\Psi\dot{m}[(h^0-h^0_o) - T_0(s-s_0)], \quad h^0 = h + \frac{C^2}{2} + gZ, \quad I_{destroyed} = T_0[\dot{S}_{gen}]$$

where Ψ_i and Ψ_o are exergy associated with mass inflow and outflows are respectively, Ψ_W is useful work done on/by system, $\dot{I}_{destroyed}$ is irreversibility of process and h^0 is the methalpy as summation of enthalpy, kinetic energy and potential energy. The other notations C is the bulk velocity of the working fluid, Z is the altitude of the stream above the sea level, g is the specific gravitational force. The irreversibility may be due to heat transfer through finite temperature difference, mixing of fluids at different temperature and mechanical friction. Exergy analysis is an effective means, to pinpoint losses due to irreversibility in a real situation.

The second law efficiency is defined as

$$\eta_{II} = \frac{\text{Actual thermal efficiency}}{\text{maximum possible (reversible) thermal efficiency}} = \frac{\text{Exergy output}}{\text{Exergy input}}$$

To analyze the possible realistic performance, a detailed exergy analysis of the coal fired thermal power plant has been carried out by ignoring the kinetic and potential energy change. For steady state flow the exergy balance for a thermal system is given as below [20]:

$$\Psi_W = \sum_{k=1}^{n}\left(1-\frac{T_0}{T_k}\right)Q_k + \sum_{k=1}^{r}[(\dot{m}\Psi)_i - (\dot{m}\Psi)_o]_k - T_0\dot{S}_{gen}$$

where Ψ_W represents the useful work done and/or by the system, the first term on the right hand side $[(1-T_0/T_k)Q_k]$ represents the exergy summation supplied through heat transfer, while changes in the exergy summation of the working fluid is represented by the second term $\sum[(\dot{m}\Psi)_i - (\dot{m}\Psi)_o]$ where i and o refers the inlet and outlet states. On the other hand, the exergy distraction and/or the irreversibility in the system is given by the last term on the right hand side, $[T_0\dot{S}_{gen}]$. The other notations such as, Q is the heat transfer rate, m is the mass flow rate of the working fluid, Ψ is the exergy flow rate per unit mass, S_{gen} is the entropy generation rate, T_0 is the ambient air temperature, T_K is the temperature of the heat source/sink at which the heat is transferred/rejected. The component wise exergy balance of the coal fire thermal power plant system is given as below.

(a) The exergy balance for boiler combustion:

The exergy balance for the combustion/boiler is give by:

$$0 = \sum_{k=1}^{r}[(\dot{m}\Psi)_{f+a} - (\dot{m}\Psi)_p]_k - T_0\dot{S}_{gen}$$

where m_{f+a} is sum of the mass of coal and air, m_p is products after combustion which gives:

$$T_0\dot{S}_{gen} = [(\dot{m}\Psi)_{f+a} - (\dot{m}\Psi)_i]$$

Entropy of the flue gas and hot products are obtained using Table A-18 to Table A-20 and Table A-27 from Cengel and Michael [19].

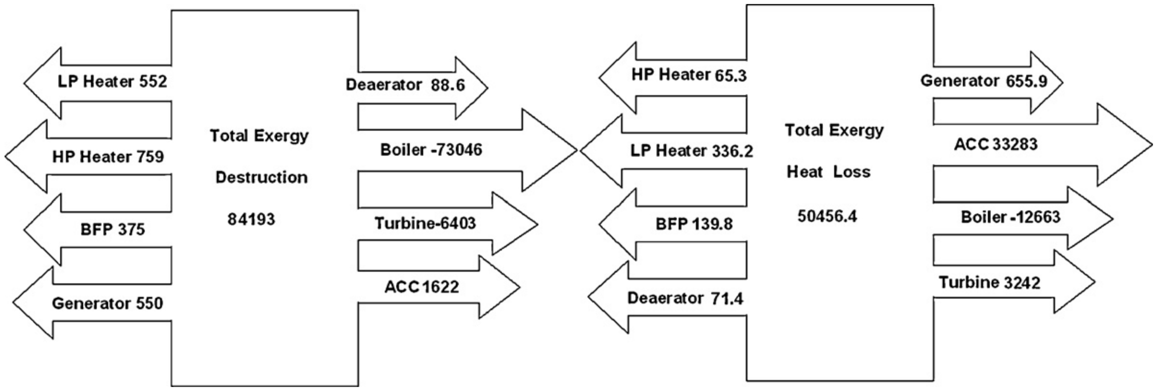

Fig. 2. Comparison of heat losses and exergy destruction (kW) in major components of the power plant.

The second law efficiency is defined as

$$\eta_{\text{III}} = \frac{\text{Exergy output}}{\text{Exergy input}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy input}}$$

$$= 1 - \frac{T_0 \dot{S}_{\text{gen}}}{(\dot{m}\Psi)_{f+a}} = \frac{(\dot{m}\Psi)_p}{(\dot{m}\Psi)_{f+a}}$$

(b) The exergy balance for high temperature heat exchanger

The exergy flow equation for the High temperature heat exchanger becomes:

$$0 = \dot{m}_p(\Psi_i - \Psi_O) - \dot{m}_w(\Psi_{10} - \Psi_9) - \dot{m}_s(\Psi_{15} - \Psi_{14}) - T_0 \dot{S}_{\text{gen}}$$

This gives:

$$T_0 \dot{S}_{\text{gen}} = [\{\dot{m}_p(h_i - h_0) - \dot{m}_w(h_{10} - h_9) - \dot{m}_s(h_{15} - h_{14})\} \\ -T_0\{\dot{m}_p(s_i - s_0) - \dot{m}_w(s_{10} - s_9) - \dot{m}_s(s_{15} - s_{14})\}]$$

The irreversibility = exergy loss is

$$\dot{I}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}}$$

The second law efficiency is:

$$\eta_{\text{II,heat}} = 1 - \frac{\dot{I}_{\text{destroyed}}}{\dot{m}_p(\Psi_i - \Psi_o)} = \frac{\dot{m}_w(\Psi_{10} - \Psi_9) + \dot{m}_s(\Psi_{15} - \Psi_{14})}{\dot{m}_p(\Psi_i - \Psi_o)}$$

Total boiler subsystem second law efficiencies is

$$\eta_{\text{II,Boiler}} = \frac{\dot{m}_w(\Psi_{10} - \Psi_9) + \dot{m}_s(\Psi_{15} - \Psi_{14})}{(\dot{m}\Psi)_f}$$

(c) The exergy balance for the high pressure turbine is give by:

$$W_{\text{HPT}} = \dot{m}_{10}(\Psi_{10} - \Psi_{11}) + (\dot{m}_{10} - \dot{m}_{11})(\Psi_{11} - \Psi_{14}) - T_0 \dot{S}_{\text{gen}}$$

This gives:

$$T_0 \dot{S}_{\text{gen}} = \dot{m}_{10}(\Psi_{10} - \Psi_{11}) + (\dot{m}_{10} - \dot{m}_{11})(\Psi_{11} - \Psi_{14}) - W_{\text{HPT}}$$

and the entropy generation rate is:

$$\dot{S}_{\text{gen}} = \dot{m}_{10}(s_{11} - s_{10}) + (\dot{m}_{10} - \dot{m}_{11})(s_{14} - s_{11})$$

The irreversibility = exergy loss is:

$$\dot{I}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0[\dot{m}_{10}(s_{11} - s_{10}) + (\dot{m}_{10} - \dot{m}_{11})(s_{14} - s_{11})]$$

The second law efficiency is:

$$\eta_{\text{II,HPT}} = 1 - \frac{\dot{I}_{\text{destroyed}}}{\dot{m}_{10}(\Psi_{10} - \Psi_{11}) + (\dot{m}_{10} - \dot{m}_{11})(\Psi_{11} - \Psi_{14})}$$

$$= \frac{W_{\text{HpT}}}{\dot{m}_{10}(\Psi_{10} - \Psi_{11}) + (\dot{m}_{10} - \dot{m}_{11})(\Psi_{11} - \Psi_{14})}$$

(d) The exergy balance for the Intermediate pressure turbine is give by:

$$W_{\text{IPT}} = \dot{m}_{15}(\Psi_{15} - \Psi_{20}) + (\dot{m}_{15} - \dot{m}_{20})(\Psi_{20} - \Psi_{23}) \\ + (\dot{m}_{15} - \dot{m}_{20} - \dot{m}_{23})(\Psi_{23} - \Psi_{19}) - T_0 \dot{S}_{\text{gen}}$$

This gives:

$$T_0 \dot{S}_{\text{gen}} = \dot{m}_{15}(\Psi_{15} - \Psi_{20}) + (\dot{m}_{15} - \dot{m}_{20})(\Psi_{20} - \Psi_{23})$$

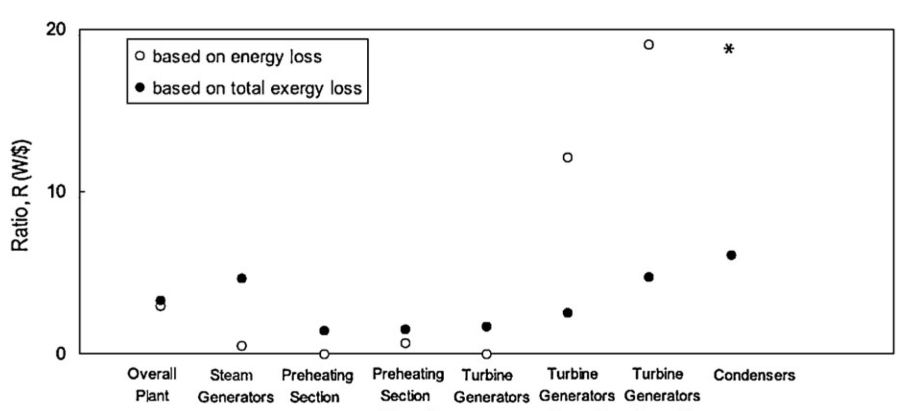

Fig. 3. Values of thermodynamic loss rate to capital cost ratio, R, for several devices in a 500 MW unit of a coal fired electrical generating station. Costs have been modified to 2002 US dollars (as explained in the text). Note that * shows R = 85:8 W/$ based on energy loss for the condenser.

$$+(\dot m_{15}-\dot m_{20}-\dot m_{23})(\Psi_{23}-\Psi_{19})-W_{\rm IPT}$$

and the entropy generation rate is:

$$\dot S_{\rm gen}=\dot m_{15}(s_{20}-s_{15})+(\dot m_{15}-\dot m_{20})(s_{23}-s_{20})+(\dot m_{15}-\dot m_{20}-\dot m_{23})(s_{19}-s_{23})$$

The irreversibility = exergy loss is:

$$\dot I_{\rm destroyed}=T_0\dot S_{\rm gen}=T_0[\dot m_{15}(s_{20}-s_{15})+(\dot m_{15}-\dot m_{20})(s_{23}-s_{20})+(\dot m_{15}-\dot m_{20}-\dot m_{23})(s_{19}-s_{23})]$$

The second law efficiency is:

$$\eta_{\rm II,IPT}=1-\frac{\dot I_{\rm destroyed}}{\dot m_{15}(\Psi_{15}-\Psi_{20})+(\dot m_{15}-\dot m_{20})(\Psi_{20}-\Psi_{23})+(\dot m_{15}-\dot m_{20}-\dot m_{23})(\Psi_{23}-\Psi_{19})}=\frac{W_{\rm IpT}}{\dot m_{15}(\Psi_{15}-\Psi_{20})+(\dot m_{15}-\dot m_{20})(\Psi_{20}-\Psi_{23})+(\dot m_{15}-\dot m_{20}-\dot m_{23})(\Psi_{23}-\Psi_{19})}$$

(e) The exergy balance for the Low pressure turbine is give by:

$$W_{\rm LPT}=\dot m_{19}(\Psi_{19}-\Psi_{27})+(\dot m_{19}-\dot m_{27})(\Psi_{27}-\Psi_{30})-T_0\dot S_{\rm gen}$$

This gives:

$$T_0\dot S_{\rm gen}=\dot m_{19}(\Psi_{19}-\Psi_{27})+(\dot m_{19}-\dot m_{27})(\Psi_{27}-\Psi_{30})-W_{\rm LPT}$$

and the entropy generation rate is:

$$\dot S_{\rm gen}=\dot m_{19}(s_{27}-s_{19})+(\dot m_{19}-\dot m_{27})(s_{30}-s_{27})$$

The irreversibility = exergy loss is:

$$\dot I_{\rm destroyed}=T_0\dot S_{\rm gen}=T_0[\dot m_{19}(s_{27}-s_{19})+(\dot m_{19}-\dot m_{27})(s_{30}-s_{27})]$$

The second law efficiency is:

$$\eta_{\rm II,LPT}=1-\frac{\dot I_{\rm destroyed}}{\dot m_{19}(\Psi_{19}-\Psi_{27})+(\dot m_{19}-\dot m_{27})(\Psi_{27}-\Psi_{30})}=\frac{W_{\rm LpT}}{\dot m_{19}(\Psi_{19}-\Psi_{27})+(\dot m_{19}-\dot m_{27})(\Psi_{27}-\Psi_{30})}$$

Condenser sub system

(f) The exergy balance for the condenser is give by:

$$0=\dot m_{30}(\Psi_{30}-\Psi_1)-\sum_{k=1}^{n}\left(1-\frac{T_0}{T_k}\right)Q_k-T_0\dot S_{\rm gen}$$

This gives:

$$T_0\dot S_{\rm gen}=\dot m_{30}(\Psi_{30}-\Psi_1)-\sum_{k=1}^{n}\left(1-\frac{T_0}{T_k}\right)Q_k$$

The irreversibility = exergy loss is:

$$\dot I_{\rm destroyed}T_0\dot S_{\rm gen}=[\{\dot m_{30}(h_{30}-h_1)\}-T_0\{\dot m_{30}(s_{30}-s_1)\}]-\sum_{k=1}^{n}\left(1-\frac{T_0}{T_k}\right)Q_k$$

The second law efficiency is:

$$\eta_{\rm II,Con}=1-\frac{\dot I_{\rm destroyed}}{\dot m_{30}(\Psi_{30}-\Psi_1)}$$

Pump sub system

(g) The exergy balance for the low pressure pump is give by:

$$-W_{\rm LPP}=\dot m_1(\Psi_1-\Psi_2)-T_0\dot S_{\rm gen}$$

This gives

The irreversibility = exergy loss is:

$$\dot I_{\rm destroyed}=T_0\dot S_{\rm gen}=\dot m_1(\Psi_1-\Psi_2)+W_{\rm LPP}=\dot m_1T_0(s_2-s_1)$$

The second law efficiency is:

$$\eta_{\rm II,LPP}=1-\frac{\dot I_{\rm destroyed}}{W_{\rm LPP}}=\frac{\dot m_1(\Psi_2-\Psi_1)}{W_{\rm LPP}}$$

(h) The exergy balance for the High pressure pump is give by:

$$-W_{\rm HPP}=\dot m_5(\Psi_5-\Psi_6)-T_0\dot S_{\rm gen}$$

This gives:

The irreversibility = exergy loss is:

$$\dot I_{\rm destroyed}=T_0\dot S_{\rm gen}=\dot m_5(\Psi_5-\Psi_6)+W_{\rm HPP}=\dot m_5T_0(s_6-s_5)$$

The second law efficiency is:

$$\eta_{\rm II,HPP}=1-\frac{\dot I_{\rm destroyed}}{W_{\rm HPP}}=\frac{\dot m_1(\Psi_6-\Psi_5)}{W_{\rm HPP}}$$

Feed water heater sub system

(i) The exergy flow equation for the high presure feed water heater (HPH1) system becomes [29]:

$$0=\dot m_{11}(\Psi_{11}-\Psi_{12})-\dot m_8(\Psi_9-\Psi_8)-T_0\dot S_{\rm gen}$$

This gives:

$$T_0\dot S_{\rm gen}=[\{\dot m_{11}(h_{11}-h_{12})-\dot m_8(h_9-h_8)\}-T_0\{\dot m_{11}(s_{11}-s_{12})-\dot m_8(s_9-s_8)\}]$$

The irreversibility = exergy loss is

$$\dot I_{\rm destroyed}=T_0\dot S_{\rm gen}$$

The second law efficiency is:

$$\eta_{\rm II,HPH1}=1-\frac{\dot I_{\rm destroyed}}{\dot m_{11}(\Psi_{11}-\Psi_{12})}=\frac{\dot m_8(\Psi_9-\Psi_8)}{\dot m_{11}(\Psi_{11}-\Psi_{12})}$$

(j) The exergy flow equation for the high presure feed water heater (HPH2) system becomes:

$$0=\dot m_{16}(\Psi_{16}-\Psi_{17})-\dot m_7(\Psi_8-\Psi_7)-T_0\dot S_{\rm gen}$$

This gives:

$$T_0\dot S_{\rm gen}=[\{\dot m_{16}(h_{16}-h_{17})-\dot m_7(h_8-h_7)\}-T_0\{\dot m_{16}(s_{16}-s_{17})-\dot m_7(s_8-s_7)\}]$$

The irreversibility = exergy loss is

$$\dot I_{\rm destroyed}=T_0\dot S_{\rm gen}$$

The second law efficiency is:

$$\eta_{\rm II,HPH2}=1-\frac{\dot I_{\rm destroyed}}{\dot m_{16}(\Psi_{16}-\Psi_{17})}=\frac{\dot m_7(\Psi_8-\Psi_7)}{\dot m_{16}(\Psi_{16}-\Psi_{17})}$$

(k) The exergy flow equation for the high presure feed water heater (HPH3) system becomes:

$$0=\dot m_{20}(\Psi_{20}-\Psi_{21})-\dot m_6(\Psi_7-\Psi_6)-T_0\dot S_{\rm gen}$$

This gives:

$$T_0\dot{S}_{gen} = [\{\dot{m}_{20}(h_{20} - h_{21}) - \dot{m}_6(h_7 - h_6)\} - T_0\{\dot{m}_{20}(s_{20} - s_{21}) - \dot{m}_6(s_7 - s_6)\}]$$

The irreversibility = exergy loss is

$$\dot{I}_{destroyed} = T_0\dot{S}_{gen}$$

The second law efficiency is:

$$\eta_{II,HPH3} = 1 - \frac{\dot{I}_{destroyed}}{\dot{m}_{20}(\Psi_{20} - \Psi_{21})} = \frac{\dot{m}_6(\Psi_7 - \Psi_6)}{\dot{m}_{20}(\Psi_{20} - \Psi_{21})}$$

(l) The exergy flow equation for the low presure feed water heater (LPH1) system becomes:

$$0 = \dot{m}_{24}(\Psi_{24} - \Psi_{25}) - \dot{m}_3(\Psi_4 - \Psi_3) - T_0\dot{S}_{gen}$$

This gives:

$$T_0\dot{S}_{gen} = [\{\dot{m}_{24}(h_{24} - h_{25}) - \dot{m}_3(h_4 - h_3)\} - T_0\{\dot{m}_{24}(s_{24} - s_{25}) - \dot{m}_3(s_4 - s_3)\}]$$

The irreversibility = exergy loss is

$$\dot{I}_{destroyed} = T_0\dot{S}_{gen}$$

The second law efficiency is:

$$\eta_{II,LPH1} = 1 - \frac{\dot{I}_{destroyed}}{\dot{m}_{24}(\Psi_{24} - \Psi_{25})} = \frac{\dot{m}_3(\Psi_4 - \Psi_3)}{\dot{m}_{24}(\Psi_{24} - \Psi_{25})}$$

(m) The exergy flow equation for the low presure feed water heater (LPH2) system becomes:

$$0 = \dot{m}_{27}(\Psi_{27} - \Psi_{28}) - \dot{m}_2(\Psi_3 - \Psi_2) - T_0\dot{S}_{gen}$$

This gives:

$$T_0\dot{S}_{gen} = [\{\dot{m}_{27}(h_{27} - h_{28}) - \dot{m}_3(h_4 - h_3)\} - T_0\{\dot{m}_{27}(s_{27} - s_{28}) - \dot{m}_2(s_3 - s_2)\}]$$

The irreversibility = exergy loss is

$$\dot{I}_{destroyed} = T_0\dot{S}_{gen}$$

The second law efficiency is:

$$\eta_{II,LPH2} = 1 - \frac{\dot{I}_{destroyed}}{\dot{m}_{27}(\Psi_{27} - \Psi_{28})} = \frac{\dot{m}_2(\Psi_3 - \Psi_2)}{\dot{m}_{27}(\Psi_{27} - \Psi_{28})}$$

(n) Dearetor sub system

It is an adiabatic mixing chamber where a hot streams 23, 22 are mixed with a cold stream 4, forming a mixture 5, the exergy supplied is the sum of the exergies of the hot and cold streams, and the exergy recovered is the exergy of the mixture. The exergy flow equation for the dearetor system becomes [30]:

$$0 = \dot{m}_{22}\Psi_{22} + \dot{m}_{23}\Psi_{23} + \dot{m}_4\Psi_4 - \dot{m}_5\Psi_5 - T_0\dot{S}_{gen}$$

where $\dot{m}_5 = \dot{m}_{22} + \dot{m}_{23} + \dot{m}_4$

This gives:

$$T_0\dot{S}_{gen} = \dot{m}_{22}\Psi_{22} + \dot{m}_{23}\Psi_{23} + \dot{m}_4\Psi_4 - \dot{m}_5\Psi_5$$

and the entropy generation rate is:

$$\dot{S}_{gen} = \dot{m}_5 s_5 - \dot{m}_{22}s_{22} - \dot{m}_{23}s_{23} - \dot{m}_4 s_4$$

The irreversibility = exergy loss is:

$$\dot{I}_{destroyed} = T_0\dot{S}_{gen} = T_0[\dot{m}_5 s_5 - \dot{m}_{22}s_{22} - \dot{m}_{23}s_{23} - \dot{m}_4 s_4]$$

The irreversibility = exergy loss is

$$\dot{I}_{destroyed} = T_0\dot{S}_{gen}$$

The second law efficiency is:

$$\eta_{II,Der} = 1 - \frac{T_0\dot{S}_{gen}}{\dot{m}_{22}\Psi_{22} + \dot{m}_{23}\Psi_{23} + \dot{m}_4\Psi_4} = \frac{\dot{m}_5\Psi_5}{\dot{m}_{22}\Psi_{22} + \dot{m}_{23}\Psi_{23} + \dot{m}_4\Psi_4}$$

Expansion valve

(o) The exergy flow equation for the expansion valve (EXP1) becomes:

$$0 = \dot{m}_{12}(\Psi_{12} - \Psi_{13}) - T_0\dot{S}_{gen}$$

This gives:

$$T_0\dot{S}_{gen} = \dot{m}_{12}[(h_{12} - h_{13}) - T_0(s_{12} - s_{13})]$$

The irreversibility = exergy loss:

$$\dot{I}_{destroyed} = T_0\dot{S}_{gen}$$

The second law efficiency is:

$$\eta_{II,EXP1} = 1 - \frac{T_0\dot{S}_{gen}}{\dot{m}_{12}(\Psi_{12} - \Psi_{13})}$$

(p) The exergy flow equation for the expansion valve (EXP2) becomes:

$$0 = \dot{m}_{17}(\Psi_{17} - \Psi_{18}) - T_0\dot{S}_{gen}$$

This gives:

$$T_0\dot{S}_{gen} = \dot{m}_{17}[(h_{17} - h_{18}) - T_0(s_{17} - s_{18})]$$

The irreversibility = exergy loss:

$$\dot{I}_{destroyed} = T_0\dot{S}_{gen}$$

The second law efficiency is:

$$\eta_{II,EXP2} = 1 - \frac{T_0\dot{S}_{gen}}{\dot{m}_{17}(\Psi_{17} - \Psi_{18})}$$

(q) The exergy flow equation for the expansion valve (EXP3) becomes:

$$0 = \dot{m}_{21}(\Psi_{21} - \Psi_{22}) - T_0\dot{S}_{gen}$$

This gives:

$$T_0\dot{S}_{gen} = \dot{m}_{21}[(h_{21} - h_{22}) - T_0(s_{21} - s_{22})]$$

The irreversibility = exergy loss:

$$\dot{I}_{destroyed} = T_0\dot{S}_{gen}$$

The second law efficiency is:

$$\eta_{II,EXP3} = 1 - \frac{T_0\dot{S}_{gen}}{\dot{m}_{21}(\Psi_{21} - \Psi_{22})}$$

(r) The exergy flow equation for the expansion valve (EXP4) becomes:

$$0 = \dot{m}_{25} - (\Psi_{25} - \Psi_{26}) - T_0\dot{S}_{gen}$$

This gives:

$$T_0\dot{S}_{gen} = \dot{m}_{25}[(h_{25} - h_{26}) - T_0(s_{25} - s_{26})]$$

The irreversibility = exergy loss:

$$\dot{I}_{destroyed} = T_0\dot{S}_{gen}$$

The second law efficiency is:

$$\eta_{\mathrm{II,EXP4}} = 1 - \frac{T_0 \dot{S}_{\mathrm{gen}}}{\dot{m}_{25}(\Psi_{25} - \Psi_{26})}$$

(s) The exergy flow equation for the expansion valve (EXP5) becomes:

$$0 = \dot{m}_{28}(\Psi_{28} - \Psi_{29}) - T_0 \dot{S}_{\mathrm{gen}}$$

This gives:

$$T_0 \dot{S}_{\mathrm{gen}} = \dot{m}_{28}[h_{28} - h_{29}) - T_0(s_{28} - s_{29})]$$

The irreversibility = exergy loss:

$$\dot{I}_{\mathrm{destroyed}} = T_0 \dot{S}_{\mathrm{gen}}$$

The second law efficiency is:

$$\eta_{\mathrm{II}} = 1 - \frac{T_0 \dot{S}_{\mathrm{gen}}}{m_{28}(\Psi_{28} - \Psi_{29})}$$

2.4. Results and discussion

The flow availabilities of different species at inlet and outlet of the Boiler have been evaluated with respect to an exergy reference thermodynamic state of P_r = 101.35 kN/m^2, T_r = 298.15 K with mole fractions of the constituents as $x_r^{O_2} = 0.2035$, $x_r^{CO_2} = 0.0003$, and $x_r^{H_2O} = 0.0303$ as recommended by Moran and Shapiro [21]. Som et al. [22] prepared *a* theoretical model of exergy balance, based on availability transfer and flow availability, in the process of pulverized coal combustion in a tubular air-coal combustor has been developed to evaluate the total thermodynamic irreversibility and second law efficiency of the process at various operating conditions. The fuel considered in the present analysis is coal, whose ultimate analysis is as follows: 70.2% C, 5.7% H, 13.4% O, 1.9% N, and 8.8% ash [23]. He noticed that as the inlet air pressure increases the second law efficiency decreases but the combustion efficiency is increases. Naterer et al. [4] presented energy and exergy analysis of subcritical boiler–turbine generator for a 32 MW coal-fired power plant. From Fig. 2, it can be observed that although the energy loss in the condenser seems higher but, the largest exergy losses occur in the boiler with the highest exergy destruction in the existing plant. This illustrates the importance of exergy analysis, as it provides the different insight and trends than that of the energy analysis. Someone performing an energy analysis would led to believe that the largest losses occur in the condenser, whereas the exergy analysis proves that they occur in the boiler.

Dincer and Rosen [24] demonstrated that, although energy and exergy values are dependent on the intensive properties of the dead state, the main results of energy and exergy analyses are usually not significantly sensitive to reasonable variations in these properties [25]. In some extreme cases, such as a rocket taking off from the ground level and flying to space, the evaluation of accurate energy and exergy values requires to be taken of care because the variations in dead-state properties are large. Saidur et al. [26] determined energy and exergy efficiencies have been determined as well. In a boiler, the energy and exergy efficiencies are found to be 72.46% and 24.89%, respectively. Dincer and Rosen [27] presented a systematic correlation appears to exist between exergy loss rate and capital cost for that purpose they have taken a thermodynamic data for the coal fired Nanticoke Generating Station consists of eight individual units, each having approximately net outputs of 500 MW. Fig. 3 shows the exergy and energy loss rate to capital cost ratio of individual components in the power plant.

It may be mentioned that so far only few studies have been done on exergy and energy analysis in coal fired thermal power plant. It is observed that in most of the cases, the major portion of exergy is lost in the combustor of a boiler. So, it should be taken into considerations for minimizing the losses in the combustion chamber. It may be due to incomplete combustion, improper insulation and entropy generation in this sub system. Table 1 shows the comparison of energy and exergy efficiencies and losses in coal fired thermal power plant with other works available in the literature.

3. Energy and exergy analysis of gas-fired combine cycle thermal power plants

Gas-fired combine cycle thermal power plant toping cycle based on the Brayton cycle and bottoming cycle based on the Rankine cycle.

3.1. Description of gas-fired combine cycle thermal power plants

The Brayton cycle was first proposed by George Brayton in 1870 for the use in the reciprocating oil-burning engine. Today, it is used for gas turbines only where both the compression and expansion processes take place in rotating machinery [28]. The total reversible processes cycle shown schematically on a T–s diagram in Fig. 4. The early gas turbines built in the 1940s and even 1950s had simple-cycle efficiencies of about 17% because of the low compressor and turbine efficiencies and low turbine inlet temperatures due to metallurgical limitations of those times [31]. Increasing the turbine inlet (or firing) temperatures this has been the primary approach taken to improve gas-turbine efficiency. The turbine inlet temperatures have increased steadily from about 540 °C (1000 °F) in the 1940s to 1425 °C (2600 °F) and even higher today [19]. We are considering for analysis cumulative gas fired combined cycle thermal power plant with all methods of the efficiency enhancement like increasing turbine inlet temperature and pressure, multy pressure heat recovery steam generator, increasing the boiler pressure, reheat and regenerative Rankine cycle, as shown in Fig. 5. The continuous mass flow diagram for one of the units of any power plant modeled in this study includes the main components such as, gas turbine (T), high, intermediate and low pressure turbine (HPT, IPT and LPT), heat recovery steam generator (HRSG), several feed water pumps (P), a dearetor, a generator (G), a condenser (COND), feed water heater, compressor (C), combustion chamber (CC). Using the balance energy and mass equations for each component in the power plant, energy, exergy flows and at each node of the plant can be calculated the numerically as well as analytically, for given set of operating conditions.

3.2. Energy analysis

To analyze the possible realistic performance, a detailed energy analysis of the gas fired combined cycle thermal power plant has been carried out by ignoring the kinetic and potential energy change. For steady state flow the energy balance for a thermal system is given as below:

$$\sum \dot{Q}_k + \dot{m}\left(h_i + \frac{C_i^2}{2} + gZ_i\right) = \dot{m}\left(h_o + \frac{C_o^2}{2} + gZ_o\right) + \dot{W}$$

where Q_k heat transfer to system from source at temperature T_k, and W is the net work developed by the system. The other notations C is the bulk velocity of the working fluid, Z, is the altitude of the stream above the sea level, g is the specific gravitational force.

The energy or first law efficiency η_{I} of a system and/or system component is defined as the ratio of energy output to the energy input to system/ component i.e.

$$\eta_{\mathrm{I}} = \frac{\text{Desired output energy}}{\text{Input energy supplied}}$$

Table 1
Comparison of energy and exergy efficiencies.

S. no	Power plants	Capacity (MW)	Exergy (second law) efficiency η_{I} (%)			Energy (first law) efficiency η_{II} (%)		Reference
			Boiler	Turbine	Overall	Boiler	Turbine	Overall
1	Yatagan	630	40.84	80.1	31.95	83.98	37.01	
2	Seyitomer	600	36.75	85.45	37.88	94.44	38.03	
3	Can	320	48.23	90.03	28.55	89.57	42.12	
4	Catalagzi	300	45.47	88.6	35.49	88.25	37.88	[1–9]
5	Kangal	457	36.45	84.19	33.09	92.10	37.19	Hasan et al. [1]
6	Afsin Elbistan	1440	39.00	86.16	25.12	93.29	42.64	
7	Orhaneli	210	45.77	64.42	26.95	85.51	37.63	
8	Soma	990	41.43	86	37.36	90.21	36.08	
9	Tuncbilek	429	44.00	86.36	31.50	89.57	38.44	
10	Zarqa	396	43.8	73.5	35.19	93.76	26.34	Aljundi [2]
11	Neyveli	50	58.62	81.2	32.46	91.9	26.91	Ganapathy et al. [6]
12	NGS	500	49.51	–	32.35	94.67	35.81	Dincer and Rosen [24]

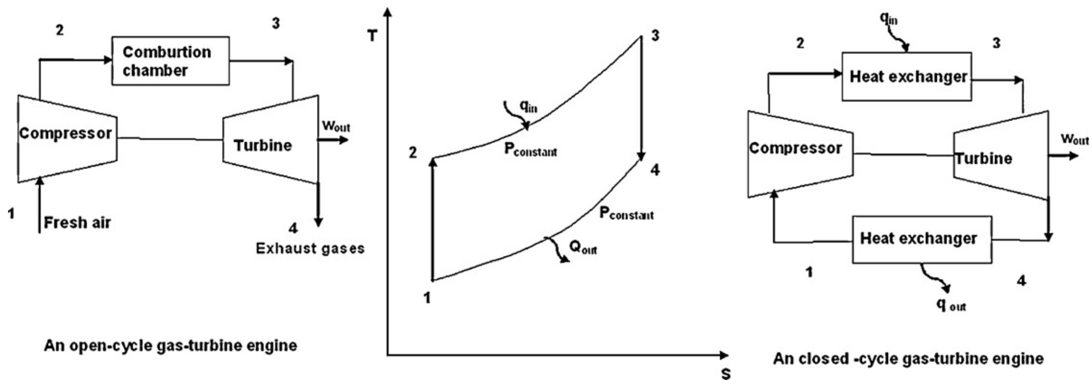

Fig. 4. Gas turbine engine.

(a) The energy balance for air compressor sub system
The energy balance for the air compressor is give by:

$$-W_C = \dot{m}_a(h_a - h_b) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = -\dot{m}_a(h_a - h_b) + W_C$$

The first law efficiency is:

$$\eta_{\mathrm{I},C} = 1 - \frac{\text{Energy loss}}{W_C} = \frac{\dot{m}_a(h_b - h_a)}{W_C}$$

(b) The energy balance for combustion chamber sub system
The energy balance for the combustion chamber is give by:

$$0 = \sum_{k=1}^{r}[(\dot{m}h)_{f+a}) - (\dot{m}h)_p)]_k - \text{Energy loss}$$

where m_{f+a} is sum of the mass of gas and air, m_p is products after combustion.
which gives:

$$\text{Energy loss} = [(\dot{m}h)_{f+a}) - (\dot{m}h)_i)]$$

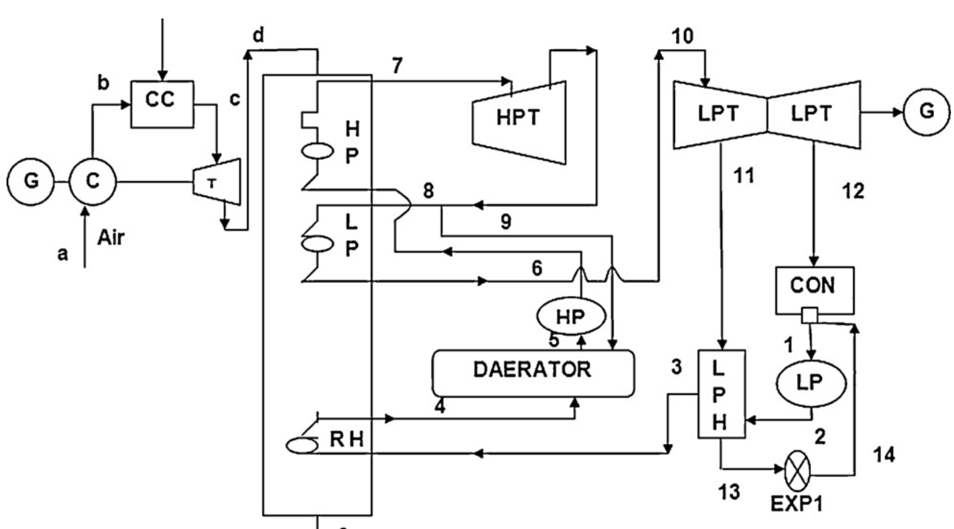

Fig. 5. Gas-fired combined cycle thermal power plant.

The first law efficiency is

$$\eta_{\mathrm{I,CC}} = \frac{\text{Energy output}}{\text{Energy input}} = 1 - \frac{\text{Energy loss}}{\text{Energy input}} = 1 - \frac{\text{Energy loss}}{(\dot{m}h)_{f+a}} = \frac{(\dot{m}h)_p}{(\dot{m}h)_{f+a}}$$

(c) The energy balance for gas turbine sub system

The energy balance for the gas turbine is give by:

$$W_{\mathrm{GT}} = \dot{m}_p(h_d - h_c) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_p(h_d - h_c) - W_{\mathrm{GT}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,GT}} = 1 - \frac{\text{Energy loss}}{\dot{m}_p(h_d - h_c)} = \frac{W_{\mathrm{GT}}}{\dot{m}_p(h_d - h_c)}$$

(d) The energy balance for heat recovery steam generator (HRSG) sub system

The energy flow equation for the boiler heat exchanger becomes:

$$0 = \dot{m}_p(h_i - h_o) - \dot{m}_4(h_4 - h_3) - \dot{m}_8(h_{10} - h_8) - \dot{m}_6(h_7 - h_6) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = [\dot{m}_p(h_i - h_o) - \dot{m}_4(h_4 - h_3) - \dot{m}_8(h_{10} - h_8) - \dot{m}_6(h_7 - h_6)]$$

The first law efficiency is:

$$\eta_{\mathrm{I,HRSG}} = 1 - \frac{\text{Energy loss}}{\dot{m}_p(h_i - h_o)} = \frac{\dot{m}_4(h_4 - h_3) + \dot{m}_8(h_{10} - h_8) + \dot{m}_6(h_7 - h_6)}{\dot{m}_p(h_i - h_o)}$$

Steam turbine sub system

(e) The energy balance for the high pressure turbine is give by:

$$W_{\mathrm{HPT}} = \dot{m}_7(h_8 - h_7) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_7(h_8 - h_7) - W_{\mathrm{HPT}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,HPT}} = 1 - \frac{\text{Energy loss}}{\dot{m}_7(h_8 - h_7)} = \frac{W_{\mathrm{HPT}}}{\dot{m}_7(h_8 - h_7)}$$

(f) The energy balance for the low pressure turbine is give by:

$$W_{\mathrm{LPT}} = \dot{m}_{10}(h_{11} - h_{10}) + (\dot{m}_{10} - \dot{m}_{11})(h_{12} - h_{11}) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_{10}(h_{11} - h_{10}) + (\dot{m}_{10} - \dot{m}_{11})(h_{12} - h_{11}) - W_{\mathrm{LPT}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,LPT}} = 1 - \frac{\text{Energy loss}}{\dot{m}_{10}(h_{11} - h_{10}) + (\dot{m}_{10} - \dot{m}_{11})(h_{12} - h_{11})} = \frac{W_{\mathrm{LPT}}}{\dot{m}_{10}(h_{11} - h_{10}) + (\dot{m}_{10} - \dot{m}_{11})(h_{12} - h_{11})}$$

(g) Condenser sub system

The energy balance for the condenser is give by:

$$0 = \dot{m}_{12}(h_{12} - h_1) - Q_k - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_{12}(h_{12} - h_1) - Q_k$$

The first law efficiency is:

$$\eta_{\mathrm{I,Con}} = 1 - \frac{\text{Energy loss}}{\dot{m}_{12}(h_{12} - h_1)}$$

Pump sub system

(h) The energy balance for the low pressure pump is give by:

$$-W_{\mathrm{LPP}} = \dot{m}_1(h_1 - h_2) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_1(h_1 - h_2) + W_{\mathrm{LPP}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,LPP}} = 1 - \frac{\text{Energy loss}}{W_{\mathrm{LPP}}} = \frac{\dot{m}_1(h_2 - h_1)}{W_{\mathrm{LPP}}}$$

(i) The energy balance for the High pressure pump is give by:

$$-W_{\mathrm{LPP}} = \dot{m}_5(h_5 - h_6) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = \dot{m}_5(h_5 - h_6) + W_{\mathrm{LPP}}$$

The first law efficiency is:

$$\eta_{\mathrm{I,HPP}} = 1 - \frac{\text{Energy loss}}{W_{\mathrm{HPP}}} = \frac{\dot{m}_5(h_6 - h_5)}{W_{\mathrm{HPP}}}$$

Feed water heater sub system

(j) The energy flow equation for the low presure feed water heater (LPH) system becomes:

$$0 = \dot{m}_{11}(h_{11} - h_{13}) - \dot{m}_2(h_3 - h_2) - \text{Energy loss}$$

This gives:

$$\text{Energy loss} = [\dot{m}_{11}(h_{11} - h_{13}) - \dot{m}_2(h_3 - h_2)]$$

The first law efficiency is:

$$\eta_{\mathrm{I,LPH}} = 1 - \frac{\text{Energy loss}}{\dot{m}_{11}(h_{11} - h_{13})} = \frac{\dot{m}_2(h_3 - h_2)}{\dot{m}_{11}(h_{11} - h_{13})}$$

(k) Dearetor sub system

The energy flow equation for the dearetor system becomes:

$$0 = \dot{m}_9 h_9 + \dot{m}_4 h_4 - \dot{m}_5 h_5 - \text{Energy loss}$$

where $\dot{m}_5 = \dot{m}_9 + \dot{m}_4$. This gives:

$$\text{Energy loss} = \dot{m}_9 h_9 + \dot{m}_4 h_4 - \dot{m}_5 h_5$$

The first law efficiency is:

$$\eta_{\mathrm{I,Dearetor}} = 1 - \frac{\text{Energy loss}}{\dot{m}_9 h_9 + \dot{m}_4 h_4} = \frac{\dot{m}_5 h_5}{\dot{m}_9 h_9 + \dot{m}_4 h_4}$$

3.3. Exergy analysis

To analyze the possible realistic performance, a detailed exergy analysis of the gas fired combined cycle thermal power plant has been carried out by ignoring the kinetic and potential energy

change. For steady state flow the exergy balance for a thermal system is given as below:

$$\Psi_W = \sum_{k=1}^{n}\left(1-\frac{T_0}{T_k}\right)Q_k + \sum_{k=1}^{r}[(\dot{m}\Psi)_i-(\dot{m}\Psi)_o]_k - T_0\dot{S}_{\text{gen}}$$

where Ψ_W represents the useful work done and/or by the system, the first term on the right hand side $[(1-(T_0/T_k))Q_k]$ represents the exergy summation supplied through heat transfer, while changes in the exergy summation of the working fluid is represented by the second term $\sum[(\dot{m}\Psi)_i-(\dot{m}\Psi)_o]$ where i and o refer the inlet and outlet states. On the other hand, the exergy distraction and/or the irreversibility in the system is given by the last term on the right hand side, $[T_0\dot{S}_{\text{gen}}]$ The other notations suchas, Q is the heat transfer rate, m is the mass flow rate of the working fluid, Ψ is the exergy flow rate per unit mass, S_{gen} is the entropy generation rate, T_0 is the ambient air temperature, T_K is the temperature of the heat source/sink at which the heat is transferred/rejected. The component wise exergy balance of the gas fire combined cycle thermal power plant system is given as below.

(a) The exergy balance for air compressor sub system

The exergy balance for the air compressor is give by:

$$-W_C = \dot{m}_a(\Psi_a-\Psi_b) - T_0\dot{S}_{\text{gen}}$$

This gives:

$$T_0\dot{S}_{\text{gen}} = \dot{m}_a(\Psi_a-\Psi_b) + W_C$$

and the entropy generation rate is:

$$\dot{S}_{\text{gen}} = \dot{m}_a(S_b-S_a)$$

The irreversibility = exergy loss is:

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}} = T_0[\dot{m}_a(S_b-S_a)]$$

The second law efficiency is:

$$\eta_{\text{II},C} = 1-\frac{\dot{I}_{\text{destroyed}}}{W_C} = \frac{\dot{m}_a(\Psi_b-\Psi_a)}{W_C}$$

(b) The exergy balance for combustion chamber sub system

$$0 = \sum_{k=1}^{r}[(\dot{m}\Psi)_{f+a}-(\dot{m}\Psi)_p]_k - T_0\dot{S}_{\text{gen}}$$

where m_{f+a} is sum of the mass of gas and air, m_p is products after combustion which gives:

$$T_0\dot{S}_{\text{gen}} = [(\dot{m}\Psi)_{f+a}-(\dot{m}\Psi)_i]$$

Entropy of the flue gas and hot products are obtained using Table A-18 to Table A-20 and Table A-27 from Cengel and Michael [19].

The second law efficiency is defined as

$$\eta_{\text{II,CC}} = \frac{\text{Exergy output}}{\text{Exergy input}} = 1-\frac{\text{Exergy loss}}{\text{Exergy input}} = 1-\frac{T_0\dot{S}_{\text{gen}}}{(\dot{m}\Psi)_{f+a}} = \frac{(\dot{m}\Psi)_p}{(\dot{m}\Psi)_{f+a}}$$

(c) The exergy balance for gas turbine sub system

The exergy balance for the gas turbine is give by:

$$W_{\text{GT}} = \dot{m}_p(\Psi_d-\Psi_c) - T_0\dot{S}_{\text{gen}}$$

This gives:

$$T_0\dot{S}_{\text{gen}} = \dot{m}_p(\Psi_d-\Psi_c) - W_{\text{GT}}$$

and the entropy generation rate is:

$$\dot{S}_{\text{gen}} = \dot{m}_p(S_c-S_d)$$

The irreversibility = exergy loss is:

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}} = T_0[\dot{m}_p(S_c-S_d)]$$

The second law efficiency is:

$$\eta_{\text{II,GT}} = 1-\frac{\dot{I}_{\text{destroyed}}}{\dot{m}_p(\Psi_d-\Psi_c)} = \frac{W_{\text{GT}}}{\dot{m}_p(\Psi_d-\Psi_c)}$$

(d) The energy balance for heat recovery steam generator (HRSG) sub system

The performance of the HRSG strongly affects the overall performance of the combined cycle power plant. HRSG is nothing but shell and tube heat exchanger, in that hot gas flow through the shell and water flow thorough tubes.

The exergy flow equation for the boiler heat exchanger becomes:

$$0 = \dot{m}_p(\Psi_i-\Psi_o) - \dot{m}_4(\Psi_4-\Psi_3) - \dot{m}_8(\Psi_{10}-\Psi_8) - \dot{m}_6(\Psi_7-\Psi_6) - T_0\dot{S}_{\text{gen}}$$

This gives:

$$T_0\dot{S}_{\text{gen}} = [\{\dot{m}_p(h_i-h_o) - \dot{m}_4(h_4-h_3) - \dot{m}_8(h_{10}-h_8) - \dot{m}_6(h_7-h_6)\} - T_0\{\dot{m}_p(s_i-s_o) - \dot{m}_4(s_4-s_3) - \dot{m}_8(s_{10}-s_8) - \dot{m}_6(s_7-s_6)\}]$$

The irreversibility = exergy loss is

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}}$$

The second law efficiency is:

$$\eta_{\text{II,HRSG}} = 1-\frac{\dot{I}_{\text{destroyed}}}{\dot{m}_p(\Psi_i-\Psi_o)} = \frac{\dot{m}_4(\Psi_4-\Psi_3)+\dot{m}_8(\Psi_{10}-\Psi_8)+\dot{m}_6(\Psi_7-\Psi_6)}{\dot{m}_p(\Psi_i-\Psi_o)}$$

Steam turbine sub system

(e) The exergy balance for the High pressure turbine is give by:

$$W_{\text{HPT}} = \dot{m}_7(\Psi_8-\Psi_7) - T_0\dot{S}_{\text{gen}}$$

This gives:

$$T_0\dot{S}_{\text{gen}} - \dot{m}_7(\Psi_8-\Psi_7) - W_{\text{HPT}}$$

and the entropy generation rate is:

$$\dot{S}_{\text{gen}} = \dot{m}_7(s_7-s_8)$$

The irreversibility = exergy loss is:

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}} = T_0[\dot{m}_7(s_7-S_8)]$$

The second law efficiency is:

$$\eta_{\text{II,HPT}} = 1-\frac{\dot{I}_{\text{destroyed}}}{\dot{m}_7(\Psi_8-\Psi_7)} = \frac{W_{\text{HPT}}}{\dot{m}_7(\Psi_8-\Psi_7)}$$

(f) The exergy balance for the Low pressure turbine is give by:

$$W_{\text{LPT}} = \dot{m}_{10}(\Psi_{11}-\Psi_{10}) + (\dot{m}_{10}-\dot{m}_{11})(\Psi_{12}-\Psi_{11}) - T_0\dot{S}_{\text{gen}}$$

This gives:

$$T_0\dot{S}_{\text{gen}} = \dot{m}_{10}(\Psi_{11}-\Psi_{10}) + (\dot{m}_{10}-\dot{m}_{11})(\Psi_{12}-\Psi_{11}) - W_{\text{IPT}}$$

and the entropy generation rate is:

$$\dot{S}_{\text{gen}} = \dot{m}_{10}(s_{10}-S_{11}) + (\dot{m}_{10}-\dot{m}_{11})(s_{11}-s_{12})$$

The irreversibility = exergy loss is:

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}} = T_0[\dot{m}_{10}(s_{10} - s_{11}) + (\dot{m}_{10} - \dot{m}_{11})(s_{11} - s_{12})]$$

The second law efficiency is:

$$\eta_{\text{II,LPT}} = 1 - \frac{\dot{I}_{\text{destroyed}}}{\dot{m}_{10}(\Psi_{11} - \Psi_{10}) + (\dot{m}_{10} - \dot{m}_{11})(\Psi_{12} - \Psi_{11})} = \frac{W_{\text{LPT}}}{\dot{m}_{10}(\Psi_{11} - \Psi_{10}) + (\dot{m}_{10} - \dot{m}_{11})(\Psi_{12} - \Psi_{11})}$$

(g) Condenser sub system

The exergy balance for the condenser is give by:

$$0 = \dot{m}_{12}(\Psi_{12} - \Psi_1) - \sum_{k=1}^{n}\left(1 - \frac{T_0}{T_k}\right)Q_k - T_0\dot{S}_{\text{gen}}$$

This gives:

$$T_0\dot{S}_{\text{gen}} = \dot{m}_{12}(\Psi_{12} - \Psi_1) - \sum_{k=1}^{n}\left(1 - \frac{T_0}{T_k}\right)Q_k$$

The irreversibility = exergy loss is:

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}} = [\{\dot{m}_{12}(h_{12} - h_1)\} - T_0\{\dot{m}_{12}(s_{12} - s_1)\}] - \sum_{k=1}^{n}\left(1 - \frac{T_0}{T_k}\right)Q_k$$

The second law efficiency is:

$$\eta_{\text{II,Con}} = 1 - \frac{\dot{I}_{\text{destroyed}}}{\dot{m}_{12}(\Psi_{12} - \Psi_1)}$$

Pump sub system

(h) The exergy balance for the low pressure pump is give by:

$$-W_{\text{LPP}} = \dot{m}_1(\Psi_1 - \Psi_2) - T_0\dot{S}_{\text{gen}}$$

This gives:

The irreversibility = exergy loss is:

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}} = \dot{m}_1(\Psi_1 - \Psi_2) + W_{\text{Lpp}} = \dot{m}_1 T_0(s_2 - s_1)$$

The second law efficiency is:

$$\eta_{\text{II,LPP}} = 1 - \frac{\dot{I}_{\text{destroyed}}}{W_{\text{Lpp}}} = \frac{\dot{m}_1(\Psi_2 - \Psi_1)}{W_{\text{LPP}}}$$

(i) The exergy balance for the High pressure pump is give by:

$$-W_{\text{HPP}} = \dot{m}_5(\Psi_5 - \Psi_6) - T_0\dot{S}_{\text{gen}}$$

This gives:

The irreversibility = exergy loss is:

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}} = \dot{m}_5(\Psi_5 - \Psi_6) + W_{\text{HPP}} = \dot{m}_5 T_0(s_6 - s_5)$$

The second law efficiency is:

$$\eta_{\text{II,HPP}} = 1 - \frac{\dot{I}_{\text{destroyed}}}{W_{\text{HPP}}} = \frac{\dot{m}_5(\Psi_6 - \Psi_5)}{W_{\text{HPP}}}$$

Feed water heater sub system

(j) The exergy flow equation for the low presure feed water heater (LPH) system becomes:

$$0 = \dot{m}_{11}(\Psi_{11} - \Psi_{13}) - \dot{m}_2(\Psi_3 - \Psi_2) - T_0\dot{S}_{\text{gen}}$$

This gives:

$$T_0\dot{S}_{\text{gen}} = [\{\dot{m}_{11}(h_{11} - h_{13}) - \dot{m}_2(h_3 - h_2)\} - T_0\{\dot{m}_{11}(s_{11} - s_{13}) - \dot{m}_2(s_3 - s_2)\}]$$

The irreversibility = exergy loss is

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}}$$

The second law efficiency is:

$$\eta_{\text{II,LPH}} = 1\frac{\dot{I}_{\text{destroyed}}}{\dot{m}_{11}(\Psi_{11} - \Psi_{13})} = \frac{\dot{m}_2(\Psi_3 - \Psi_2)}{\dot{m}_{11}(\Psi_{11} - \Psi_{13})}$$

(k) Dearetor sub system

It is an adiabatic mixing chamber where a hot streams 9 is mixed with a cold stream 4, forming a mixture 5, the exergy supplied is the sum of the exergies of the hot and cold streams, and the exergy recovered is the exergy of the mixture. The exergy flow equation for the dearetor system becomes:

$$0 = \dot{m}_9\Psi_9 + \dot{m}_4\Psi_4 - \dot{m}_5\Psi_5 - T_0\dot{S}_{\text{gen}}$$

where $\dot{m}_5 = \dot{m}_9 + \dot{m}_4$. This gives:

$$T_0 S_{\text{gen}} = \dot{m}_9\Psi_9 + \dot{m}_4\Psi_4 - \dot{m}_5\Psi_5$$

and the entropy generation rate is:

$$\dot{S}_{\text{gen}} = \dot{m}_5 s_5 - \dot{m}_9 s_9 - \dot{m}_4 s_4$$

The irreversibility = exergy loss is:

$$\dot{I}_{\text{destroyed}} = T_0 S_{\text{gen}} = T_0[\dot{m}_5 s_5 - \dot{m}_9 s_9 - \dot{m}_4 s_4]$$

The irreversibility = exergy loss is

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}}$$

The second law efficiency is:

$$\eta_{\text{II,Der}} = 1 - \frac{T_0\dot{S}_{\text{gen}}}{\dot{m}_9\Psi_9 + \dot{m}_4\Psi_4} = \frac{\dot{m}_5\Psi_5}{\dot{m}_9\Psi_9 + \dot{m}_4\Psi_4}$$

Expansion valve

(l) The exergy flow equation for the expansion valve (EXP) becomes:

$$0 = \dot{m}_{13}(\Psi_{13} - \Psi_{14}) - T_0\dot{S}_{\text{gen}}$$

This gives:

$$T_0\dot{S}_{\text{gen}} = \dot{m}_{13}[(h_{13} - h_{14}) - T_0(S_{13} - S_{14})]$$

The irreversibility = exergy loss:

$$\dot{I}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}}$$

The second law efficiency is:

$$\eta_{\text{II,EXP1}} = 1 - \frac{T_0\dot{S}_{\text{gen}}}{\dot{m}_{13}(\Psi_{13} - \Psi_{14})}$$

3.4. Results and discussion

Sue and Chuang [15] performed exergy analysis based on load variation. The exergy loss at 50% load is three times that of 100% load due to the lower steam pressure in the HRSG. Therefore, the plant selected combined cycle plant operating efficiency at 100% load is 2.4% higher than at 50% load. Khaliq and Kaushik [17] presented second-law efficiency of gas fire thermal power plant varying the number of reheat process and compression ratio in gas turbine. The first-law efficiency of the adiabatic turbine increases with the increase in pressure ratio. The second-law efficiency decreases with the pressure ratio, but increases with the cycle temperature ratio since a greater proportion of the available work lost at the higher temperature may be recovered. The exergy destruction in the reheat turbine increases with the pressure ratio, the number of reheat stages and the pressure drop in each re-heater first-law

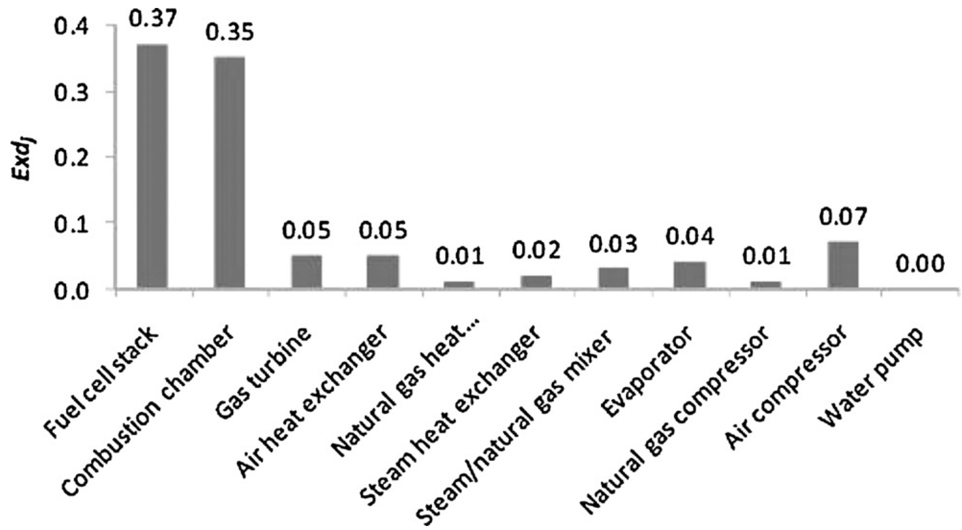

Fig. 6. Exergy destructions per mole of methane consumed for the devices in the integrated system.

and second-law efficiencies of the combined cycle increases up to the pressure ratio of 32, and then they start decreasing with increases in the pressure ratio. But it is interesting to note that the second-law efficiency of the combined cycle is greater than that of the first-law efficiency for same pressure-ratio. If the pressure ratio is too low, then the gas-turbine cycle and combined-cycle efficiencies and their specific work-outputs drop, whereas the steam cycle work-output increases due to the high gas-turbine exhaust temperature. If the pressure ratio is too high, the compressor and turbine works increase but their difference, the net gas-turbine work output drops. Franco and Russo [33] analyzed the heat recovery steam generator (HRSG), as a first step in the analysis of the whole plant. They handle this problem adopting both a thermodynamic and a thermoeconomic objective function instead of the usual pinch point method. Thermodynamic optimization has the purpose to diminish energy losses, expressed on exergy basis, while the aim of the thermoeconomic optimization is the minimization of the cost function associated with the system/plant, sum of the cost of exergy inefficiencies and the cost of the HRSG. Proposed methods have been applied to some HRSG configurations, including some present commercial plants. The results of the application of the thermoeconomic optimization lead to a meaningful increase of the thermal efficiency of the plant that approaches the 60%. Dincer et al. [34] reported energy and exergy assessments of integrated power generation using solid oxide fuel cells (SOFCs) with internal reforming and a gas turbine cycle. The other main exergy destruction is attributable to electrochemical fuel oxidation in the SOFC. The energy and exergy efficiencies of the integrated system reach 70–80%, which compares well to the efficiencies of approximately 55% typical of conventional combined-cycle power generation systems. Variations in the energy and exergy efficiencies of the integrated system with operating conditions are provided, showing, for example, that the SOFC efficiency is enhanced if the fuel cell active area is augmented. The SOFC stack efficiency can be enhanced by reducing the steam generation while increasing the stack size. Fig. 6 shows exergy destructions per mole of methane consumed for the devices in the integrated system. Ertesvag et al. [35] presented a concept for natural-gas (NG) fired power plants with CO_2 capture was investigated based on the exergy analysis. Natural gas was reformed in an auto-thermal reformer (ATR), and the CO_2 was separated before the hydrogen-rich fuel was used in a conventional combined-cycle (CC) process. The main purpose of the study was to investigate the integration of the reforming process and the combined cycle. An increase of the turbine in let temperature (TIT) from 1250 to 1350 °C and1450 °C increased the net electric-power production to 50.6% and 52.2%, respectively, of the NG for the plant with reforming and CO_2 capture. The corresponding results for the conventional combustion chamber (CC) plant were 60.2% and 61.0% respectively [36]. For the plant with reforming and CO_2 capture, a combination of 1450 °C TIT and ATR product-feed heat exchange gave a net electric-power production of 53.3% of the NG. Kanoglu and Dincer [39] studied the performance assessment of various cogeneration systems through energy and exergy efficiencies. The cogeneration plants considered include steam-turbine system, gas-turbine system, diesel-engine system, and geothermal system, and the results of the analysis are given in Table 2. Reddy and Mohamed [11] determined gas turbine main combustion chamber as the major source of exergy destruction rate. The exergy destruction rate in the main combustion chamber is found be very high as compared to other parts

Table 2
Energy and exergy analyses results for four different cogeneration systems.

S. no	Parameters	Steam co-generation	Gas-turbine cogeneration	Diesel-engine cogeneration	Geothermal cogeneration
1	Inlet temperature of hot fluid to heater (°C)	249	303	400	100
2	Exit temperature of hot fluid from heater (°C)	212	211	111	8
3	Inlet temperature of water to heater (°C)	50	50	50	50
4	Exit temperature of water from heater (°C)	90	90	90	90
5	Net power output, $\dot{w}_{net}$ (kW)	10,000	10,000	20,000	10,000
6	Heating supplied, Q_{heat} (kW)	13,500	13,500	13,500	13,500
7	Exergy input to the plant Ψ_{in}(kW)	50.960	52,000	45,620	26,650
8	Total exergy destruction in the plant, $\dot{I}_{dr}$ (kW)	39,200	40,240	23,860	14,885
9	Energy efficiency, η_{cogen} (%)	47.8	46.8	78.2	16.1
10	Exergy efficiency, η_{cogen} (%)	23.1	22.6	47.7	44.1

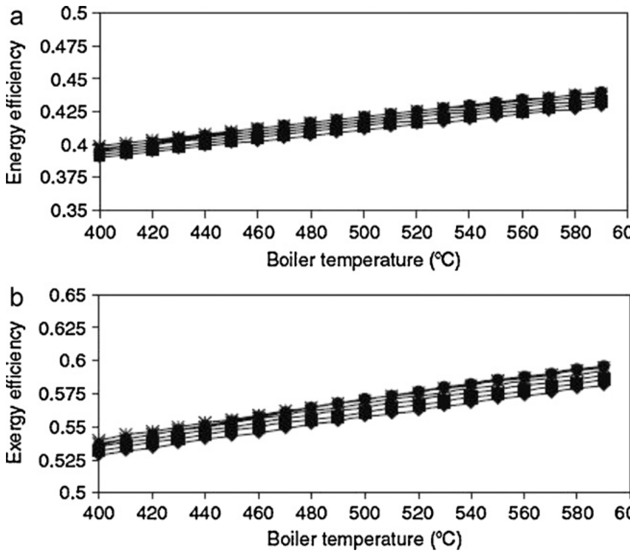

Fig. 7. Variation of (a) energy efficiency and (b) exergy efficiency with boiler temperature for various boiler pressures ((♦) 10 MPa, (■) 11 MPa, (▲) 12 MPa, (×) 13 MPa, (*) 14 MPa, (●) 15 MPa) for the Ghazlan Power Plant.

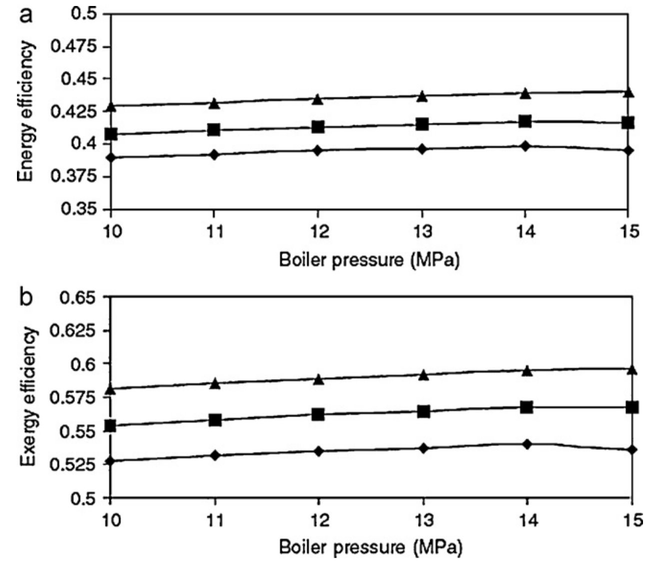

Fig. 8. Variation of (a) energy efficiency and (b) exergy efficiency with boiler pressure for various boiler temperatures ((♦) 400 °C, (■) 500 °C, (▲) 590 °C)) for the Ghazlan Power Plant.

of the system. Whereas the higher pressure ratio, results in an increase in exergy destruction rate in the gas turbine cycle components [40,41]. For the same pressure ratio, the combined cycle net work output increases with higher turbine inlet temperatures. The exergy destruction rate in the combustion chambers and the gas turbine cycle components reduces with higher gas turbine inlet temperature.

4. Improvement potentials

Dincer and Rosen [46] presented exergy improvement methods for coal fired thermal power plants. They have taken thermodynamic parameters from the Ghazlan Power Plant in Saudi Arabia [42]. With the boiler pressure of 12.5 MPa and the temperature is 510 °C. while the condenser pressure of 50.8 mmHg, temperature of 38.4 °C. Whereas the regenerator pressure of 132 kPa. Habib et al. [43] have indicated that for maximum efficiency of a single-reheat cycle, the reheat pressure should be approximately 19% of the boiler pressure. In the simulation they varied the inlet temperature of the high-pressure turbine (or the outlet temperature of the boiler) between 400 °C and 600 °C with an interval of 10 °C. For each temperature, the pressure has been changed from 10 to 15 MPa with an interval of 1 MPa. The pressure at the inlet of the low-pressure turbine was taken to be 20% that of the high-pressure turbine. In Fig. 7a and b show the energy and exergy efficiency has been plotted against the boiler temperature for values between 400 °C and 600 °C, for different boiler pressures of 10, 11, 12, 13, 14 and 15 MPa respectively. All energy efficiency profiles increase almost linearly with the boiler temperature and the energy efficiencies vary from 38% to 45%. However, the exergy efficiency varies between 52.5% and 60%. Energy and exergy efficiency are plotted against boiler pressure in Fig. 8a and b for three different temperatures (400 °C, 500 °C and 590 °C). Although both energy and exergy efficiencies increase slightly with increasing boiler pressure. The increase must be weighed against the added cost of equipment to increase the pressure.

The maximum energy efficiency for the three curves occurs at a boiler pressure of 14 MPa. Therefore, this pressure can be considered as a thermodynamic optimum for such a cycle under the design conditions. Although the difference between the profiles is larger at lower temperatures like 400 °C, they approach each other as temperature increases and tend to match near 600 °C. Kelly et al. [47] analyzed the irreversibilities (exergy destruction) within a component of an energy conversion system emphasizing that the irreversibility associated with a component can be represented in two parts. The first part depends on the inefficiencies of the considered component while the second part depends on the system structure and the inefficiencies of the other components of the overall system. Thus, the exergy destruction occurring within a component can be split into two parts: (a) endogenous exergy destruction exclusively due to the performance of the component being considered and (b) exogenous exergy destruction caused also by the inefficiencies within the remaining components of the overall system. The paper discussed four different approaches developed by the author [48] for calculating the endogenous part of exergy destruction as well as the approach based on the structural theory. The advantages, disadvantages and restrictions for applications associated with each approach have been presented. Kotas [32] explained about the mismatching of heat capacities of heat transfer media, considering the heat transfer taking place in a parallel-flow mode and/or when the heat capacities of the streams are mismatched in a counter-flow heat exchanger. Even when the temperature difference is very small at one end of the heat exchanger, there will still be appreciable irreversibility rate due to heat transfer over a finite temperature difference at other points in the heat exchanger. This type of intrinsic irreversibility is associated with the particular physical configuration of the plant [37,38].

5. Conclusions

Exergy analysis is shown in this article to be able to help understand the performance of coal fired, gas fired combined cycle thermal power plants and identify design possible efficiency improvements. It gives logical solution improving the power production opportunities in thermal power plants [44,45]. By the exergy analysis we can conclude that main energy loss in boiler in coal based thermal power plant and combustion chamber in gas fired combined cycle thermal power plant. Of course, in every plant component such as a boiler, combustion chamber there is some intrinsic irreversibility which cannot, owing to the present state of technological development, be eliminated. In addition,

exergy methods are useful in assessing which improvements are worthwhile, and should be used along with other pertinent information to guide efficiency improvement efforts for steam power plants. Of course, Efficiency of some plant components is improved by increasing their size. For example, heat exchangers of a given design perform better when the heat transfer areas are increased. However, this involves extra cost and hence there is a limiting size beyond which further increase would not be justified economically.

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